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1.2 Load Analysis
1.2.2 Real Time Investigations

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DELIVERY D22 (Workpackage 3)

Task 1.2 Load Analysis
Subtask 1.2.2 Real Time Investigations

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Modifications

Revision	Date	Modifications
A	09/29/2003	First Issue
B	07/30/2004	Seismic Loads

Summary

The document contributes to the 5th Framework Programme HTR-E Workpackage 3, Delivery D22 – Load analysis. It describes the results of the subtask “Real time investigations”.

The dynamic simulation was accomplished with simulation tool *MLDyn*. The modules of *MLDyn* were parameterized for GT-MHR turbomachine rotor layout.

With parameterized simulation tool *MLDyn* calculations of AMB control loop behavior with axial and radial AMB’s were performed at:

- start up
- speed levels and unbalances
- static loads
- dynamic loads (dynamic stiffness)
- seismic loads

The analysis of dynamic loads was carried out with the simulation-based method for the AMB Design regard to the parameter Dynamic Stiffness (DDS). In the design phase the method is applicable to the theoretical proof of the reliability performance of active magnetic bearing systems.

List of Symbols

Symbols

e	eccentricity
F	force
G_z	disturbance transfer function
i	current
K	gain
n	rotor speed
ω	frequency
s	Laplace operator
S	stiffness
t	time
u	voltage
x,y,z	rotor displacement in x-, y-, z-direction

Indexes

1...6	No. of radial magnetic bearing
al	allowable
d	dynamic
D	design, deviation moment
Dis	disturbance
E	deviation moment
imb	imbalances
L	bearing level
M	measurement level
max	maximum
min	minimum
n	necessary
Op	operational
R	controller
Res	reserve
S	sensor
set	setpoint
$static$	static
x,y,z	in x-, y-, z-direction

Contents

1. Introduction/Objectives

The document contributes to the 5th Framework Programme HTR-E Workpackage 3, Delivery D22 – Load analysis. The task is subdivided into two subtasks:

1. Modal and harmonic analysis (performed by NRG)
2. Real time investigations (performed by University Zittau/Goerlitz)

The first objective of the task is to analyse the loads at machines which are comparable with HTR rotating components (GT-MHR and PBMR) with Modal and Harmonic response analyses and Transient dynamic analyses based on Finite Element Codes. The numerical results provide the boundary and loading conditions for the design and the simulation tool (*MLDyn*).

The second objective is to perform real time investigations of the complete system with the modular simulation tool *MLDyn* (unbalances, loads, speed levels). A first design will be examined with the DDS-Method (Design on Dynamic Stiffness).

This document describes the results of the dynamic simulation calculations of the AMB (active magnetic bearing) supported turbomachine shaft under the following conditions:

- start up
- speed levels and unbalances
- static loads
- dynamic loads (dynamic stiffness)
- seismic loads

The used inputs are based on the following documents:
Framatome ANP (FRA): HTR-E – AMB and CB – Functional Requirements
HTR-E-02/06-D-3-1-1
Societe de Mechanique Magnetique (S2M): HTR-E Magnetic Bearing Concept Proposal
HTR-E 03/12 D-3-2-1-1
NRG: HTR-E Load Analysis – Modal and Harmonic Analysis HTR-E-04/03-D-3-1-2-1

As defined by all partners in the workpackage 3 (2nd WP meeting in Zittau) the configuration to be investigated is the GT-MHR reactor type.

2. Dynamic simulation

2.1 Simulation tool *MLDyn*

The dynamic simulation calculations were performed with the simulation tool *MLDyn*. *MLDyn* was especially developed for theoretical investigations in the field of active magnetic bearing systems. It contains all components of the AMB control loop (Fig. 1).

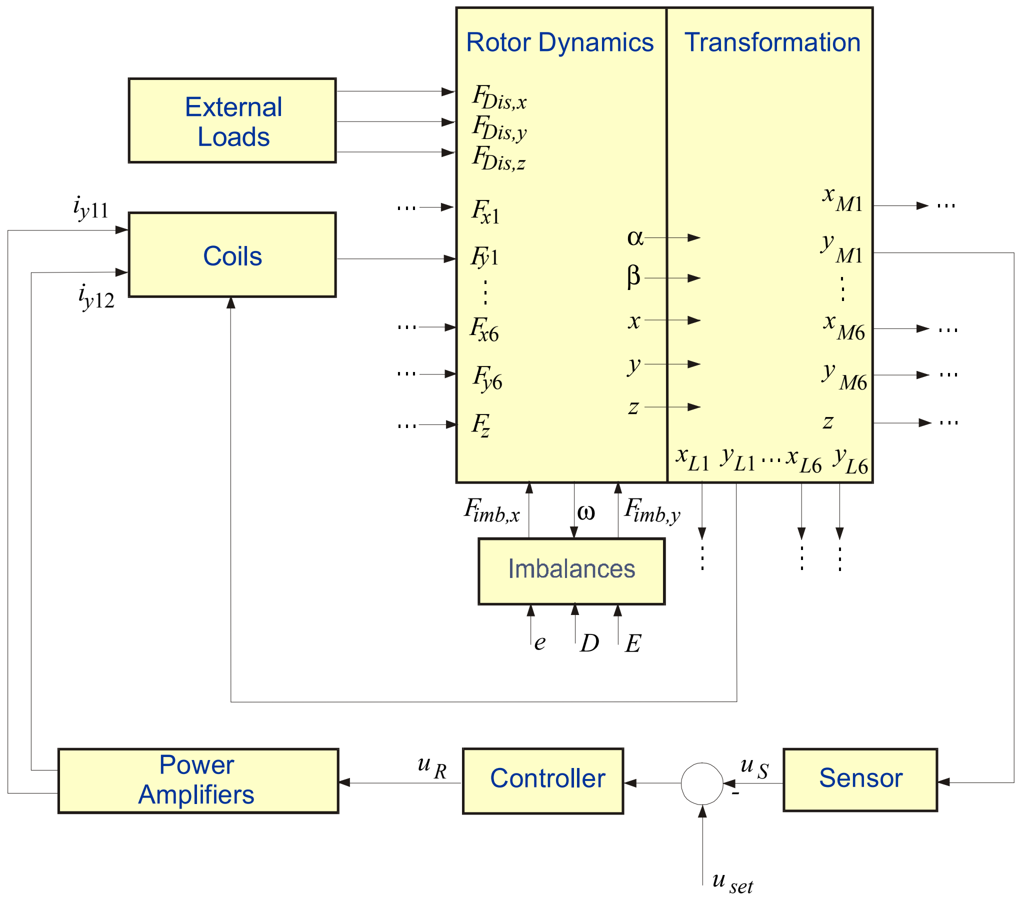


Fig. 1: Strucure of the simulation tool *MLDyn*

The simulation tool is characterized by
- modelling the magnetic bearing control loop for completely active magnetically supported rigid rotors,
- a modular type of construction and an easy exchange of components of the control loop,
- emergency operation for imbalances and unit loads,
- the possibility of configuration for any magnetic bearing systems by adjustment of characteristic parameters and structures,
- verification at pilot plants.

Application fields are
- investigations of the dynamics of rotors,
- loop investigations (transients, numbers of revolutions, critical operating situations, start-up and shut-down operations),
- investigations of the control loop stability,
- support in controller design,
- preparation of experiments.

2.2 Parametrization

As required, the turbomachine shaft of GT-MHR reactor type is supported by 6 radial and 2 axial AMB’s. Regarding the FRA requirements and the S2M magnetic bearing concept the simulation tool has been parameterized. Fig. 2 shows the radial and axial bearing location on the turbomachine shaft.

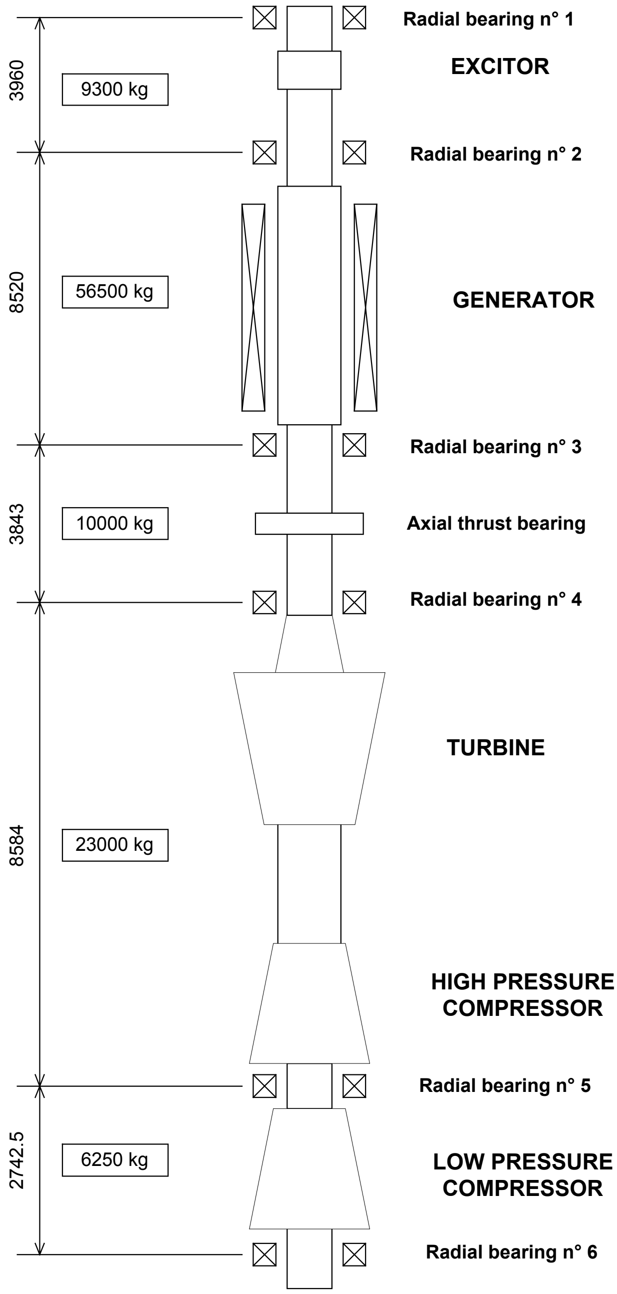


Fig. 2: Bearing location on turbomachine shaft (origin: Framatome ANP)

Table 1 gives an overview about the given and calculated main parameters of the rotor dynamics module.

Table 1: Parameter list of rotor dynamics module (*MLDyn*)

PARAMETER	VALUE	COMMENT
Mass of Rotor	104,610 kg	NRG calculation
Axial Moment of Inertia	4.254 $*10^4$ kgm^2	NRG calculation
Radial Moment of Inertia	5.309 $*10^6$ kgm^2	NRG calculation
Centre of Gravity	16.16 m	NRG calculation
Rotor Speed Nominal Speed Maximum Speed	 3000 rpm 3600 rpm	FRA requirements
Eccentricity	10 µm	FRA requirements
Location of AMB radial AMB 1 radial AMB 2 radial AMB 3 radial AMB 4 radial AMB 5 radial AMB 6 axial AMB	 27.75 m 23.79 m 15.27 m 11.43 m 2.84 m 0.60 m 13.88 m	FRA requirements

Hereafter the the modules „sensor“, „controller“, „amplifier“ and „coils“ were parameterized.

Table 2: Parameter list of modules „sensor“, „controller“, „amplifier“ and „coils“ (*MLDyn*)

MODULE	PARAMETER	VALUE		COMMENT
SENSOR	gain time constant	10V/mm 1ms or less		assumed
CONTROLLER	gain derivative time integration time	variabel 25 ms 250 ms		depend. on operational state
AMPLIFIER	gain closed loop voltage max. current bias current	6 A/V 500 V 60 A 18 A		S2M specification variabel
COILS	 magnetic surface magnetic length No. of windings material active length time constants (at nominal gap)	radial 0.01396 m^2 ~ 355 mm 43/Coil V130-35A 158 mm (6t) 316 mm(12t) 0,0583 s (6t) 0,1540 s (12t)	axial 1.38 m^2 474...561mm 134/Coil 0,8125 s	S2M specification

Based on the input parameter of the coils (S2M specification) the force-current-gap characteristic fields of the radial and axial magnets are calculated for the module „coils“ of the simulation tool (Fig. 3 and 4). The radial AMB's have a load capacity of 60,000 N resp. 120,000 N at nominal air gap and maximum current of 60 A (FRA requirement). The starting capacity is 32,000 N resp. 64,000 N at maximum air gap (at catcher bearing position).

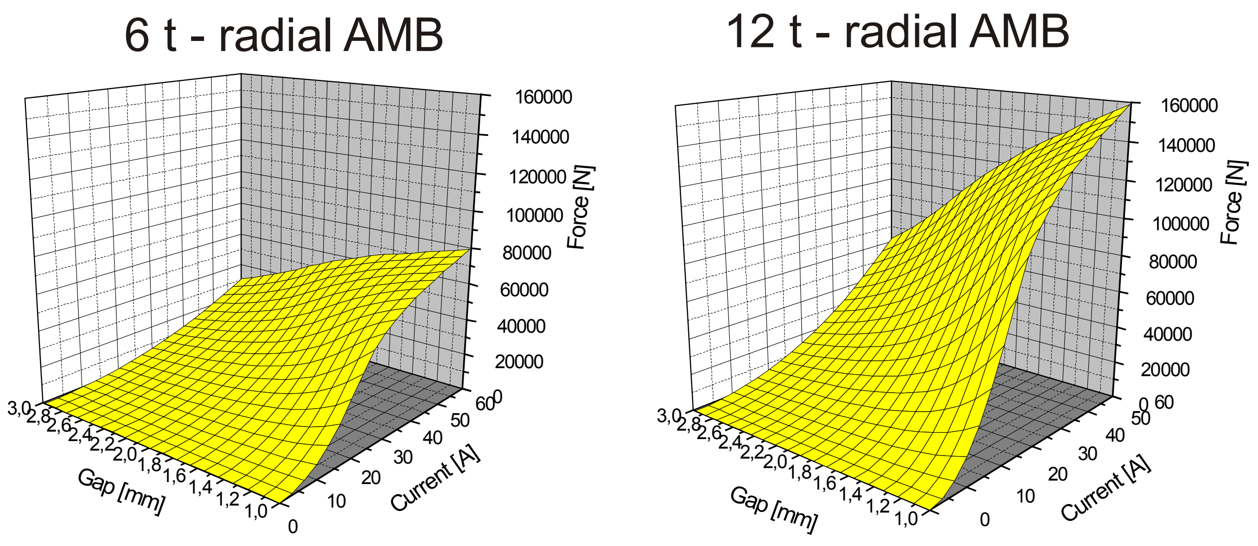


Fig. 3: Characteristic fields of radial AMB's (one quadrant)

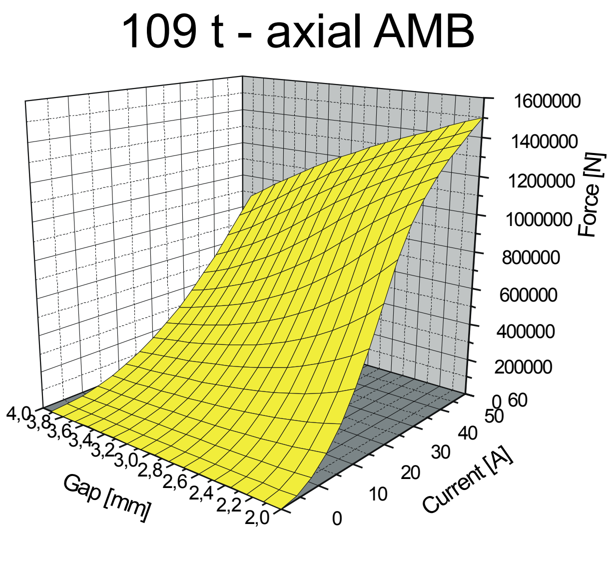


Fig. 4: Characteristic field of axial AMB (4 coils design of S2M)

To realize the lift up of the rotor it is neccessary to have a double trusk disk design, i.e. two axial AMB's with 2x 1,090,000 N load capacity at nominal air gap. The starting capacity at maximum air gap (lower catcher bearing position) is then 2x 850,000 N.

2.3 Dynamic simulation calculations

With parameterized simulation tool *MLDyn* calculations of AMB control loop behavior with axial and radial AMB's were performed at:

- start up
- speed levels and unbalances
- static loads
- dynamic loads (dynamic stiffness)
- seismic loads

These calculation are necessary to evaluate the functionality and reliability of the designed AMB's.

2.3.1 Start up

Radial AMB's

At the start-up operation the shaft must be lifted from an arbitrary position in the catcher bearings to the setpoint position (mainly center position). A minimal overshoot and a short settling time are required.

In Fig. 5 the overshoot of the rotor position at all radial AMB's is shown during the start-up. The overshoot depends on the adjusted controller gain K_R – increased gain results in minimal overshoot. In the investigated range of controller gain the radial AMB's are able to start up the rotor.

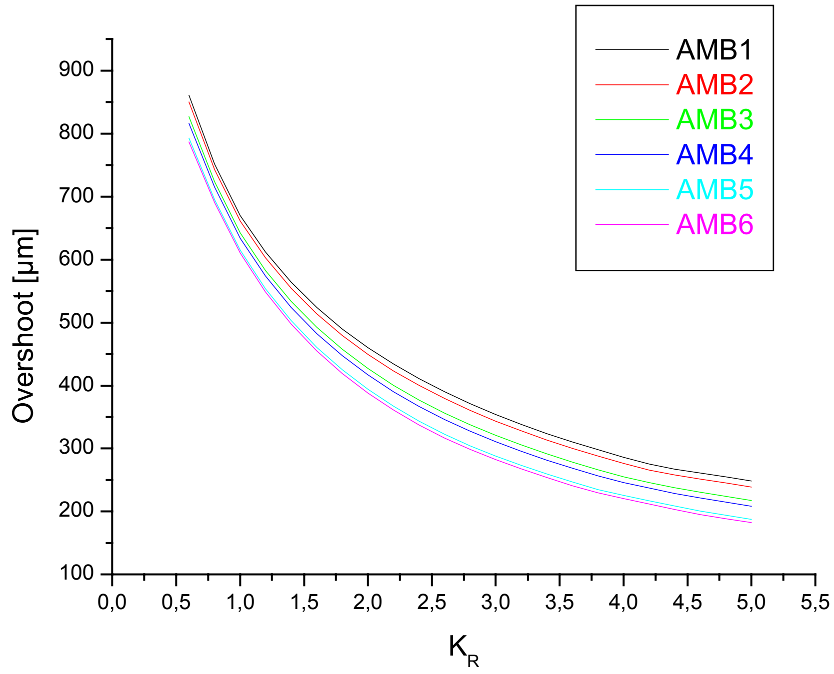


Fig. 5: Overshoot vs. Controller Gain for radial AMB 1...6

Axial AMB

At the start-up operation the shaft must be lifted from the lower position in the catcher bearings to the setpoint position (mainly center position). In the investigated range of controller gain no overshoot occur. Fig. 6 shows the settling time vs. the controller gain. Settling time has a maximum at K_R =1.5. With higher gain the settling time decreases. The axial bearing is able to start up the rotor.

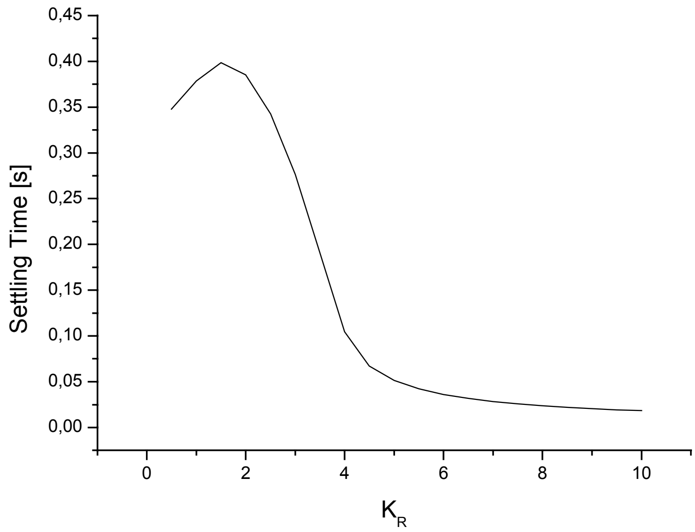


Fig. 6: Settling Time vs. Controller Gain for axial AMB

2.3.2 Speed levels and unbalances

High loads at the shaft can occur due to resonances vibrations at critical speeds. The reason for these vibrations are unbalances. The unbalances are caused by the eccentricity of the center of gravity and deviation moments. The eccentricity e = 10 µm is given by the FRA requirements.
The arising force is proportionally to the square of the rotational speed.

Fig. 7 shows the maximum amplitude of rotor position and the bearing force vs. the controller gain and the rotational speed exemplarily for the radial AMB 1.

NOTE: The amplitude of rotor position and the bearing force for AMB 1...6 can be seen on Annex 1.

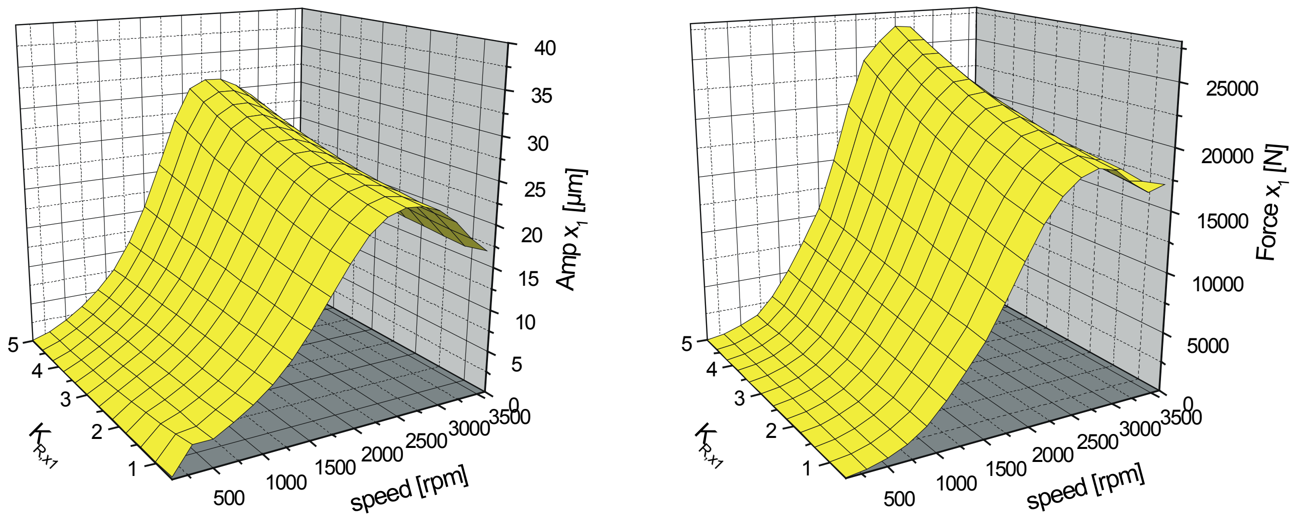


Fig. 7: Amplitude of rotor position and bearing force vs. speed and controller gain for radial AMB 1

The amplitude of rotor position varies with the controller gain and the rotational speed. Increased amplitudes occur at the nominal speed of 3,000 rpm but the radial AMB's are able to react the loads caused by unbalances up to full speed (3,600 rpm).

Fig. 8 shows the result of investigations to determine an optimal controller gain where the amplitude of rotor position is minimized at all rotational speed. In the next tasks (2.1.2 Concept of optimized controller algorithms) it is planned to apply controllers where the gain is adapted dependent on the speed.

Up to 800 rpm a high gain is necessary to minimize the amplitude of rotor position. At higher speed the gain must be decreased.

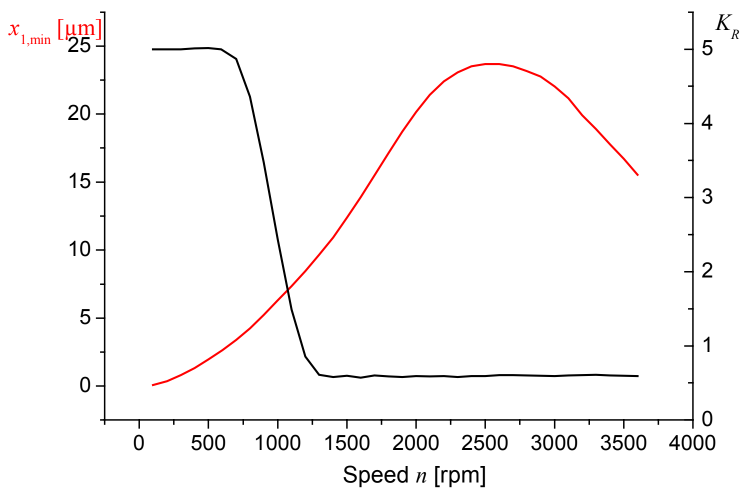


Fig. 8: Minimum Amplitude of Rotor Position for Full Speed Range and Optimal Gain for Radial AMB 1

2.3.3 Static loads

Radial AMB's

According the FRA requirements a static load of 8500 N is acting between radial AMB 2 and 3. The operational speed is 3000 rpm. By variation of controller gain the amplitude of rotor position and the bearing forces were investigated. Fig. 9 shows the results for radial AMB 2.

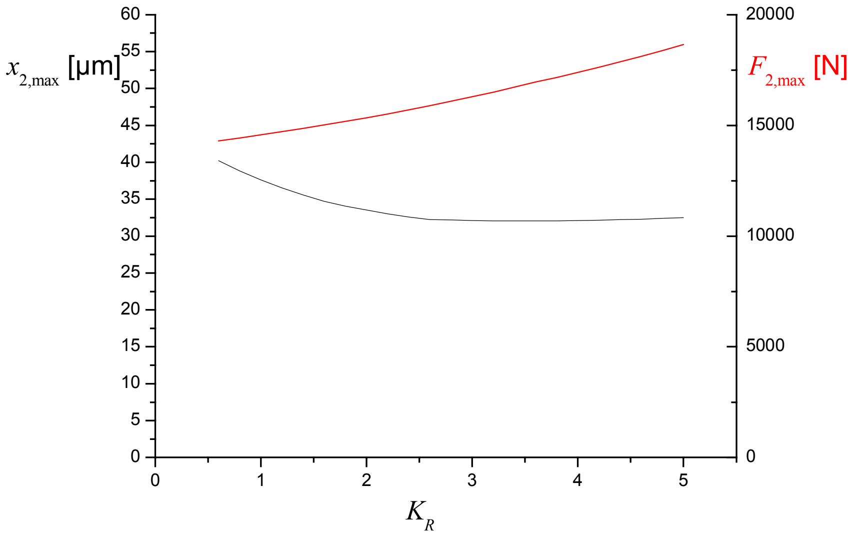


Fig. 9: Amplitude of Rotor Position and Bearing Force vs. Controller Gain for radial AMB 2

The amplitude decreases with higher controller gain but therefore higher bearing forces are necessary (higher stiffness).
In the full range of gain the radial AMB's are able to react the static loads.

NOTE: The amplitude of rotor position and the bearing force for AMB 1...6 can be seen on Annex 2.

Axial AMB

Static loads are caused by the load from the turbomachine rotor mass and aerodynamical axial forces (from turbine and compressors). The axial load amounts 639238 N (NRG calculation). Fig. 10 shows the amplitude of rotor position and the bearing force vs. the variable controller gain for the axial AMB.

The amplitude decreases with higher controller gain but therefore higher bearing forces are necessary (higher stiffness).
In the full range of gain the axial AMB is able to react the static loads.

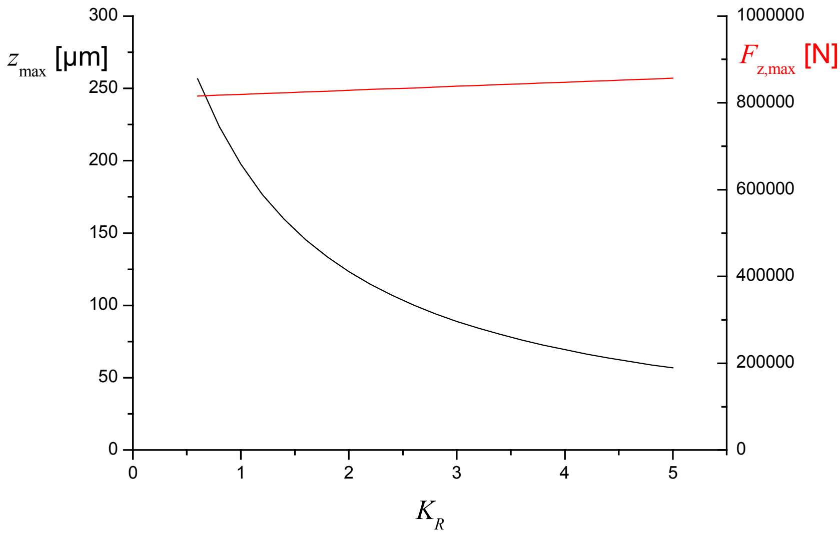


Fig. 10: Amplitude of Rotor Position and Bearing Force vs. Controller Gain for axial AMB

2.3.4 Dynamic loads – Design on dynamic stiffness (DDS)

Radial AMB's

For the design of the AMB's it is necessary to consider the behavior at dynamic loads acting on the shaft. Based on the static loads an amplitude of the disturbance force of 8500 N is used for the investigation. Additionally the frequency of the load is changed between 0 and 500 Hz and the controller gain between 0.6 and 5.0. Fig. 11 shows the bearing force vs. the freqency of the load and the controller gain for radial AMB 2.

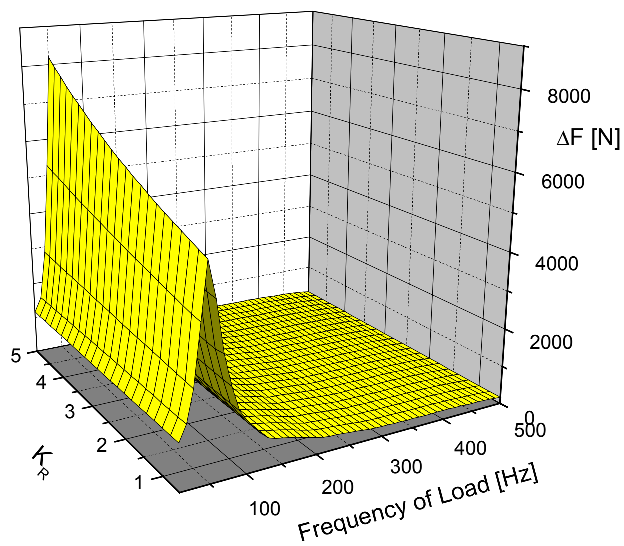


Fig. 11: Bearing Force vs. Freqency of Load and Controller Gain for radial AMB 2

As seen on Fig. 10 high forces occur at 50 Hz that corresponds to the nominal speed of 3000 rpm.

Design regard to the parameter dynamic stiffness (DDS Method)

The basis for the optimal design is firstly the knowledge about the rotor dynamic behaviour during operation and secondly the specific loads acting on the rotor.
In the design phase the simulation-based method for the AMB Design regard to the parameter Dynamic Stiffness (DDS) is applicable to the theoretical proof of the reliability performance of active magnetic bearing systems. This is based on the Simulation Tool *MLDyn*. Necessary dynamic stiffness is introduced as a criterion for reliability and allowable rotor displacement as a criterion for quality.

Analogous to conventional bearings at magnetic bearings load capacity and stiffness can be defined. The static load capacity F_{max} is the maximum load capacity for a static force over an unlimited time period. It is stated by

- maximal forces of the application
- the geometry of the arrangement
- the design of the machine

The load is limited by the maximal receivable ampere turns, i.e.restricted by coil temperature and voltage limit. The stiffness S is given by the negative position stiffness K_S of the electromagnet and the controller gain.

Different from the conventional bearing the AMB - stiffness is dependent on the operating frequency ω of the disturbance forces. Therefore it there is a difference between static and dynamic stiffness.
The dynamic stiffness for a closed loop is

$$S_d = \frac{1}{G_z(s)} = \frac{F_{Dis}(s)}{X(s)} \tag{1}$$

where
s - Laplace-Operator
$G_z(s)$ - disturbance transfer function
$F_{Dis}(s)$- disturbance force dependent on frequency
$X(s)$ - displacement

The static stiffness is the limit of S_d for $t \to \infty$ and $s \to 0$ by a static force ΔF_{static} different from the working point.

$$S_{static} = \lim_{s \to 0} S_d(s) = \frac{\Delta F_{static}}{X(s)} \tag{2}$$

The attainable maximum stiffness is dependent on the frequency behaviour of the control loop. The proof of the reliability performance requires an analysis of the whole frequency spectrum of the disturbance forces.

Depend on the expected application requirements, the necessary AMB force F_D is determined. This is followed by an adjustment of AMB components like sensors and amplifier and a first optimization of the control loop. Based on the required AMB feature, a necessary stiffness S_n is defined, which the bearing has to be adapted to:

$$S_n = \frac{F_{\max}}{X_{al}} \tag{3}$$

where $F_{\max} = F_D + F_{Res}$

F_D - design force
F_{Res} - reserve force (safety allowance)
X_{al} - allowable rotor displacement

The allowable rotor displacement X_{al} is given by constructive and operative (positioning accuracy) requirements of the application and is introduced as the quality criterion, which has to be secured by the AMB system.
The accessible dynamic stiffness is calculated by *MLDyn* in consideration to the selected AMB design:

$$S_d(\omega) = \frac{F_D + F_{Dis}(\omega)}{X(\omega)} \tag{4}$$

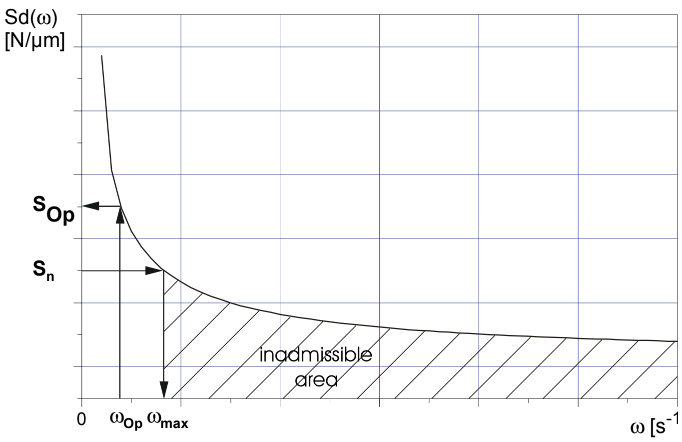


Fig. 12: Closed loop dynamic stiffness vs. frequency

Fig. 12 is a qualitative representation of $S_d(\omega)$. The result at the point of intersection between the simulated dynamic stiffness and the necessary stiffness S_n is the maximum frequency $\omega_{\max}$ for acting disturbance forces. This frequency limits the allowable operating range and therefore it is the design limit. Above the design limit ($\omega > \omega_{max}$) the dynamic stiffness is below the necessary stiffness. The perfect operation of the drafted AMB is not guaranteed. On the basis of an analysis of the real frequency spectrum at a machine the dominating frequency ω_{Op} is determined as well as the operating stiffness S_{Op}. The criterion of the reliability performance of the AMB is

$$\boldsymbol{S_{Op} > S_n} \tag{5}$$

If the relationship (5) is not fulfilled in a first step a controller fine tuning follows. If the tuning is not successful in a next step the AMB parameter must be determined again. The whole algorithm is shown in Fig. 13.

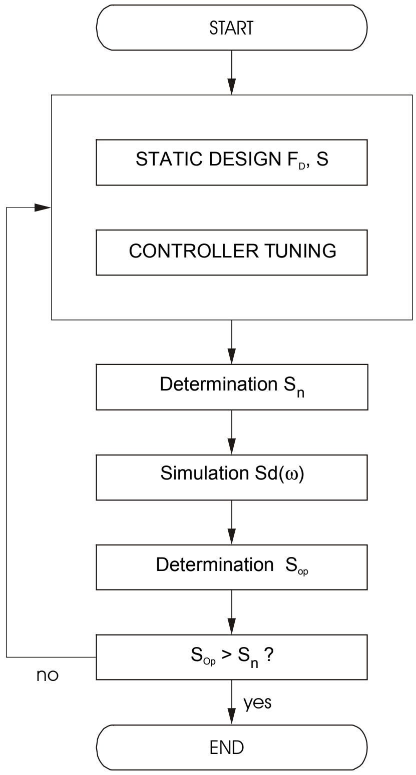


Fig. 13: Algorithm of the DDS method to proof of the reliability performance for AMB

To avoid contact with the catcher bearings not more than 80% of radial gap between catcher bearing and rotor are allowed. The allowable rotor displacement X_{al} as quality criterion is assumed with 400 µm.

The design force F_D of both types of radial AMB's is $6*10^4$ N (for 6t bearing) resp. $12*10^4$ N (for 12t bearing). With a reserve of 10% the maximum force F_{max} amounts $6.6*10^4$ N resp. $13.2*10^4$ N.

Then the necessary stiffness S_n yields **165 N/µm** resp. **330 N/µm**.

Fig. 14 shows the stiffness S_d (ω) exemplarily for AMB 2 (12t bearing) which was calculated with the simulation tool *MLDyn*.

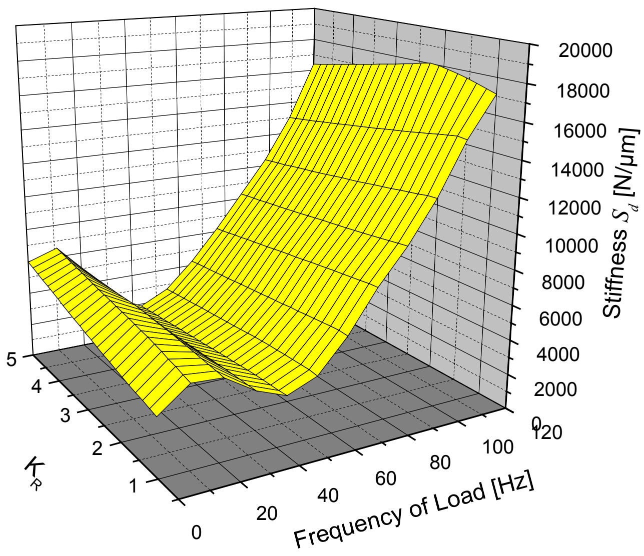


Fig. 14: Dynamic Stiffness vs. Freqency of Load and Controller Gain for radial AMB 2

Resonance frequencies are expected in the range 0-110 Hz (FRA requirements and NRG calcultions), i.e. ω_{max} is 110 Hz. The operating frequency ω_{op} is in the range below ω_{max}.

The minimal operating stiffness $S_{op,min}$ in the range between 0-110Hz amounts 1861 N/µm and is higher then S_n = 330 N/µm.

Therefore the criteria $\boldsymbol{S_{Op}} > \boldsymbol{S_n}$ is fulfilled in the operating range and for all controller gains K_R.

NOTE: The bearing force and the dynamic stiffness for AMB 1...6 can be seen on Annex 3.

Table 3 shows the necessary and the minimum operating stiffness for all radial AMB's. The criteria $\boldsymbol{S_{Op}} > \boldsymbol{S_n}$ is fulfilled for all bearings.

Table 3: Necessary and the minimum operating stiffness for radial AMB's

radial AMB	necessary stiffness S_n	minimum operating stiffness $S_{Op,\,min}$
1	165 N/µm	1025 N/µm
2	330 N/µm	1861 N/µm
3	330 N/µm	1627 N/µm
4	165 N/µm	973 N/µm
5	165 N/µm	513 N/µm
6	165 N/µm	456 N/µm

Axial AMB

The loads depend on the aerodynamical forces in the range 0-60 Hz. The controller gain is changed between 0.5 and 5.0.
Fig. 15 shows the bearing force vs. the freqency of the load and the controller gain for the axial AMB.

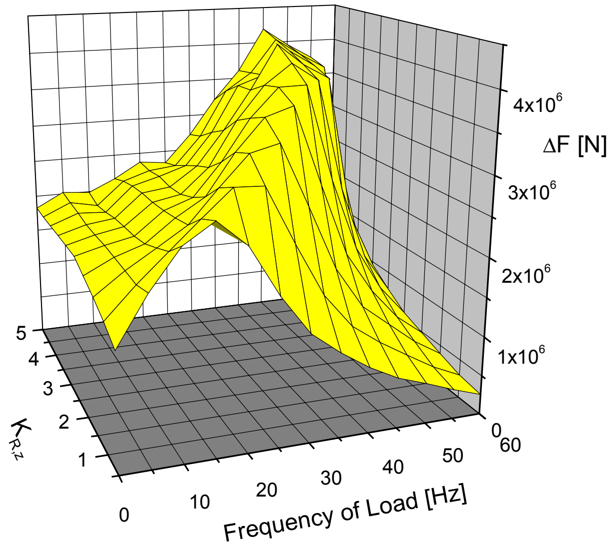


Fig. 15: Bearing Force vs. Freqency of Load and Controller Gain for axial AMB

As seen on Fig. 14 high forces occur between 20 to 50 Hz depending on the controller gain. For minimal bearing forces a small gain should be used.

For the DDS-Method the allowable rotor displacement X_{al} must be fixed. Using 80% of axial gap between catcher bearing and rotor disk X_{al} amounts 400 µm.

The design force F_D of axial AMB is 2x1.090 $*10^6$N. With a reserve of 10% the maximum force $F_{\max}$ amounts 2.398$*10^6$ N.

Then the necessary stiffness S_n yields **5995 N/µm**.

Fig. 16 shows the stiffness S_d (ω) for the axial AMB which was calculated with the simulation tool *MLDyn*.

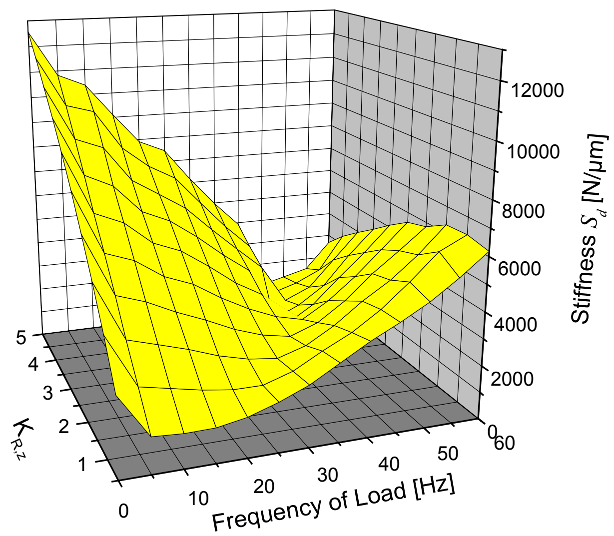


Fig. 16: Dynamic Stiffness vs. Freqency of Load and Controller Gain for axial AMB

Resonance frequencies are expected in the range 0-60 Hz (FRA requirements and NRG calcultions), i.e. ω_{max} is 60 Hz. The operating frequency ω_{op} is in the range below ω_{max}.

The minimal operating stiffness $S_{op,min}$ in the range between 0-60Hz amounts 1011N/µm and is **lower** then S_n = 5995 N/µm.

Therefore the criteria $\boldsymbol{S_{Op} > S_n}$ is not fulfilled in the operating range and for all controller gains K_R.

The axial AMB needs to be further optimized regarding the time constants of the coils (inductivity) to meet the criteria of the DDS method. A way to decrease the time constants can be the development and usage of more powerful power amplifiers (higher closed loop voltage).

2.3.5 Seismic loads

Radial AMB‘s

Based on the input parameters calculated by NRG – see Document HTR-E-04/03-D-3-1-2-1 – the dynamic behavior of the TM shaft in case of an earthquake has been simulated.
Table 4 shows the seismic loads in radial and axial direction acting on the 6 radial AMB’s and the axial AMB. The frequency of these loads has been varied between 0 and 100 Hz. In addition the dynamic behavior was investigated without and with rotational speed (3000 rpm) as well overspeed (3600 rpm).

Table 4: Seismic loads in x- and y-direction (NRG calculation)

radial AMB	seismic load x-direction	seismic load y-direction
1	5922 N	19957 N
2	11553 N	102970 N
3	6817 N	137240 N
4	1343 N	75560 N
5	343 N	57158 N
6	89 N	14182 N

By variation of controller gain, the amplitude of rotor position and the bearing forces were investigated. Fig. 17 shows the results for radial AMB 3 which has to compensate the highest seismic loads.

Due to the high loads in y-direction it isn't possible to suspend the rotor in the magnetic bearings in case of an earthquake. The seismic loads are higher then the load capacity of the magnetic bearings – especially for radial bearing 3 and 4.

In x-direction the loads are smaller and the amplitudes of rotor position are not higher then 150 µm (maximum at radial bearing 1). The maximum displacement occurs at small gains and lower load frequencies (5 Hz). For minimal amplitudes the controller gain must be increased.

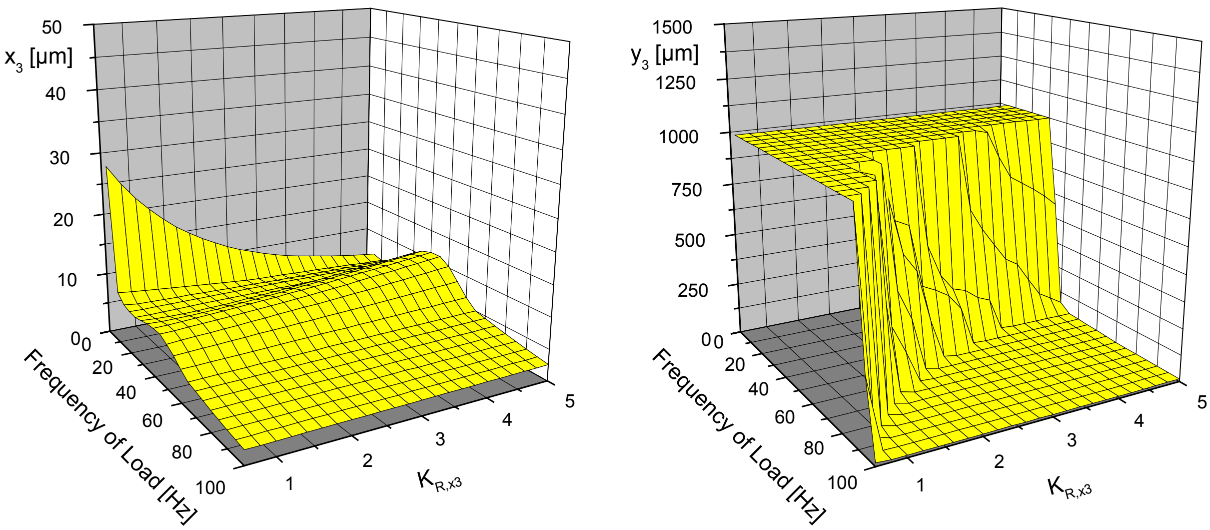


Fig. 17: Amplitude of Rotor Position in x- and y-direction vs. Freqency of Load and Controller Gain for radial AMB 3 (n = 0 rpm)

At nominal speed and overspeed additional loads are caused by the unbalance of the shaft.

In y-direction it is also not possible to suspend the rotor in the magnetic bearings as the loads are higher then the load capacity of the AMB‘s.
For the x-direction the compensation of the seismic and unbalance loads is possible.
The maximum displacement of the shaft running at 3000 rpm is not higher then 330 µm (maximum at radial bearing 6) depending on the controller gain.
The maximum displacement of the shaft running at 3600 rpm is not higher then 370 µm (maximum at radial bearing 5) depending on the controller gain.
For minimal amplitudes the controller gain must be increased.

The Fig. 18 and 19 show the results of simulation for radial AMB 3 in case of nominal speed and overspeed.

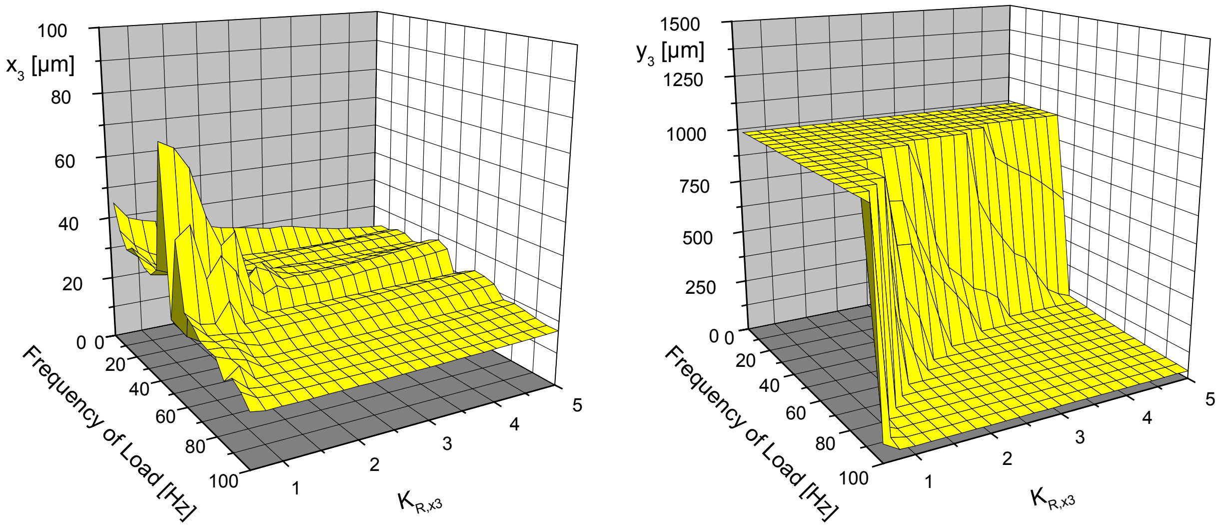


Fig. 18: Amplitude of Rotor Position in x- and y-direction vs. Freqency of Load and Controller Gain for radial AMB 3 (n = 3000 rpm)

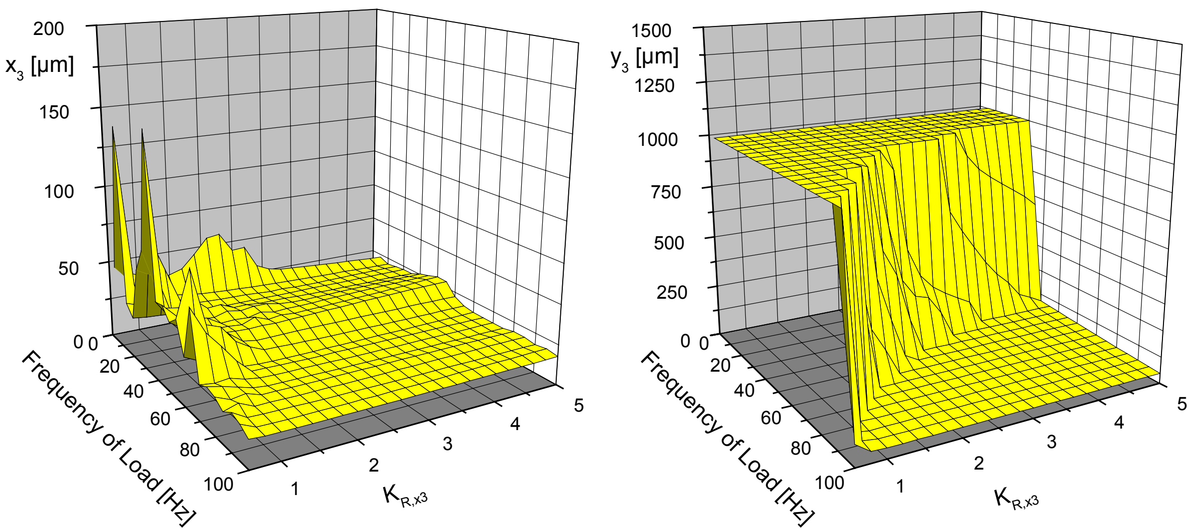


Fig. 19: Amplitude of Rotor Position in x- and y-direction vs. Freqency of Load and Controller Gain for radial AMB 3 (n = 3600 rpm)

NOTE: The amplitude of rotor position for AMB 1...6 can be seen on Annex 4.

Axial AMB

The seismic load in axial direction was also calculated by NRG. It amounts 258,670 N. For simulation of the dynamic behavior this load was oscillating in a frequency range between 0 and 100 Hz.

By variation of controller gain, the amplitude of rotor position and the bearing forces were investigated. Fig. 20 shows the results for the axial AMB.

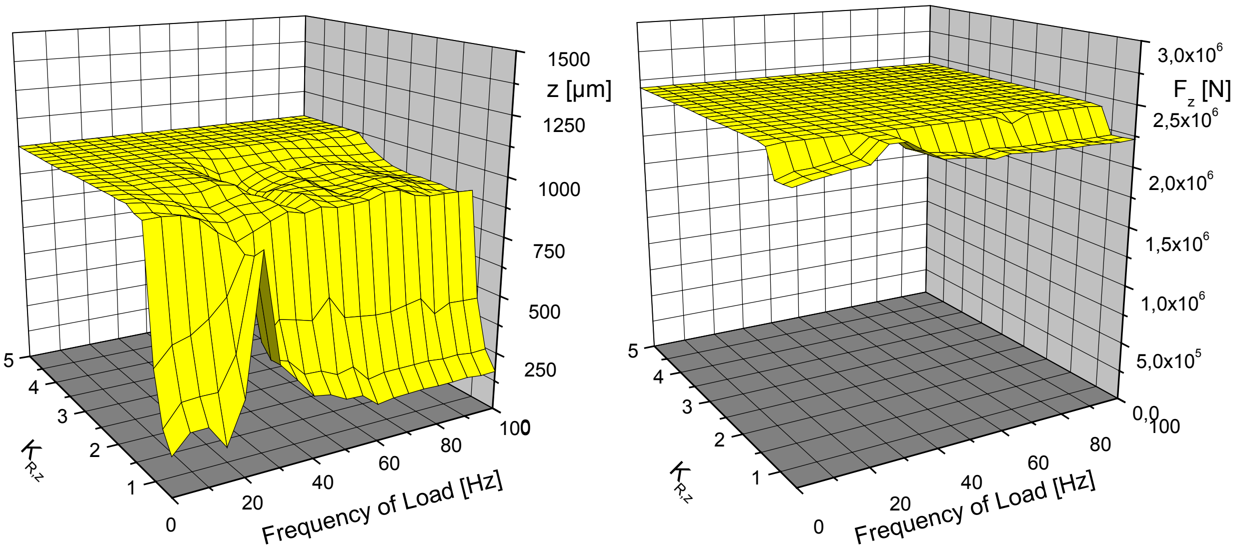


Fig. 20: Amplitude of Rotor Position and Bearing Force in z-direction vs. Freqency of Load and Controller Gain for axial AMB

The amplitude of rotor position varies with the controller gain and the frequency of the seismic load. Increased amplitudes (max 870 µm) occur at a load frequency of 25 Hz.

With small gains i.e. smaller stiffness the axial AMB is able to react the seismic load but the bearings forces are at the limit of the load capacity.

3. Conclusions

Regarding the subtask „Real Time Investigations“ dynamic simulation calculations were performed to evaluate the behaviour of the magnetic bearings support.
The simulation tool MLDyn was parameterized for all control loops of radial and axial AMB‘s of the GT-MHR turbomachine shaft.
To simulate the behaviour in case of loads acting on the shaft the following operational and disturbance situations were calculated:
- start up
- speed levels and unbalances
- static loads
- dynamic loads (dynamic stiffness)
- seismic loads

Results:

	Radial AMB‘s		
Investigation	Parameters	Criteria	Results
Start-up	Start at CB position, controller gain 0.6...5.0	Overshoot	increased gain → minimal overshoot, radial AMB’s able to start rotor
Speed levels and unbalances	Speed 0...3000 rpm, controller gain 0.6...5.0, Eccentricity 10 µm	Amplitude of rotor position, bearing forces	increased amplitudes and bearing forces at nominal speed (3000 rpm), radial AMB’s able to react the loads up to full speed (3600 rpm), minimal displacement achievable with variable gain of feedback controller
Static loads	load 8500 N const., speed 3000 rpm, controller gain 0.6...5.0	Amplitude of rotor position, bearing forces	amplitude decreases with higher controller gain, but higher bearing forces necessary (higher stiffness), radial AMB’s able to react the static loads
Dynamic loads	load 8500 N const., frequency 0...500 Hz, controller gain 0.6...5.0	Bearing force, Dynamic stiffness	high bearing forces at 50 Hz DDS method: criteria fulfilled (operating stiffness>necessary stiffness)

Seismic loads	load different at the 6 radial AMB's, also different in x- and y-direction, frequency 0...100 Hz, Controller gain 0.6...5.0, speed: 0, 3000, 3600 rpm	Amplitude of rotor position, bearing forces	radial AMB's able to react the static loads in x-direction (amplitude decreases with higher controller gain), radial AMB's not able to react the static loads in y-direction (loads higher then bearing load capacity)

	Axial AMB		
Investigation	Parameters	Criteria	Results
Start-up	Start at CB position, Controller gain 0.5...10.0	Settling time	no overshoot, settling time maximum at controller gain=1.5, higher gain then 1.5 → decreased settling time, axial AMB able to start rotor
Static loads	load 639,238 N const. Controller gain 0.5...5.0	Amplitude of rotor position, bearing forces	amplitude decreases with higher controller gain, but higher bearing forces necessary (higher stiffness), axial AMB able to react the static loads
Dynamic loads	load 0...767872N variable (aerodynamical forces), frequency 0...60 Hz, Controller gain 0.6...5.0	Bearing force, Dynamic stiffness	high bearing forces at 50 Hz DDS method: criteria not fulfilled (operating stiffness<necessary stiffness)
Seismic loads	load 258670 N const., frequency 0...60 Hz, Controller gain 0.6...5.0		axial AMB able to react the seismic load with small controller gains (small stiffness), bearing forces on the limit

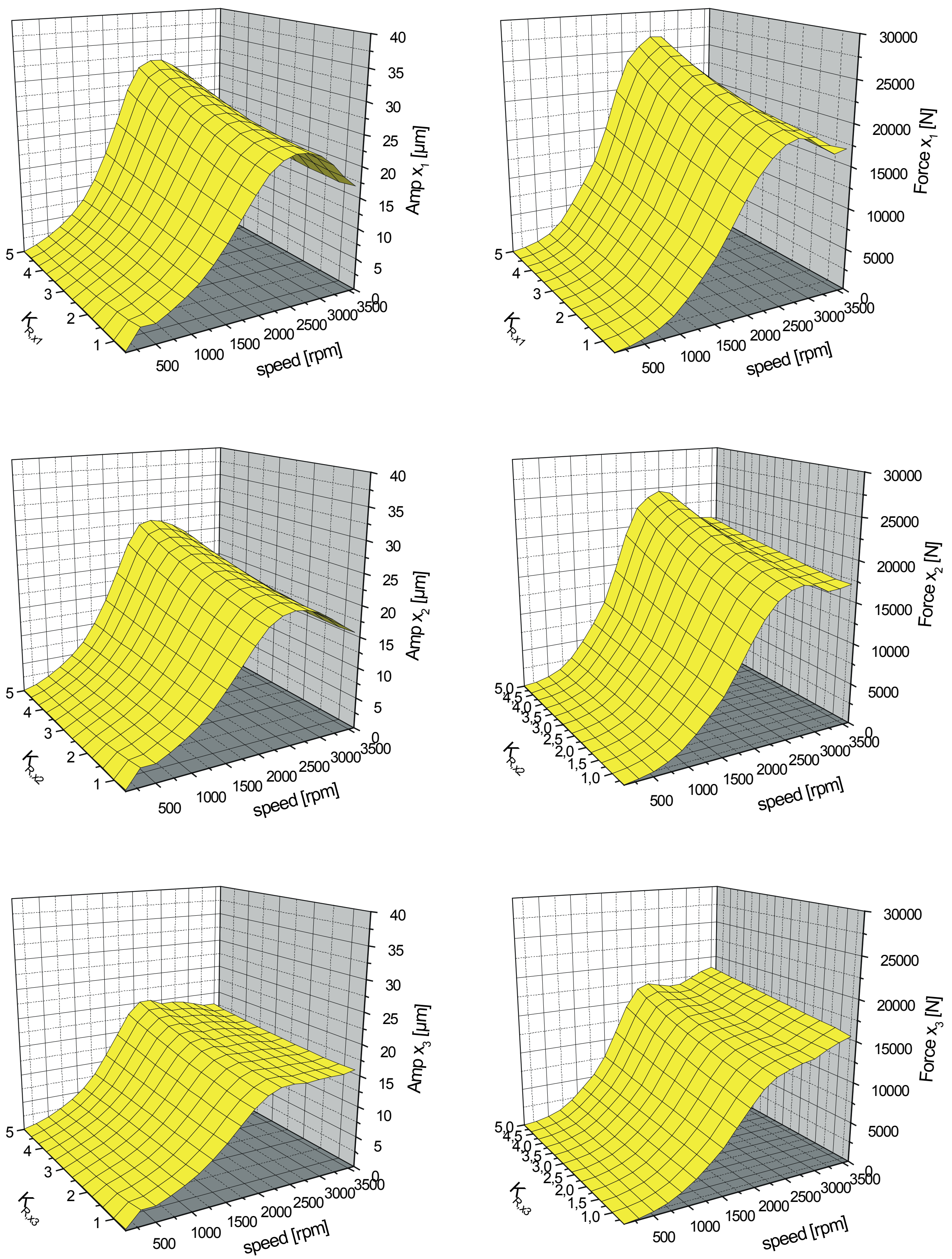
Recommendations:

The radial AMB's can be used as designed for the support of the turbomachine shaft since they are able to control the loads at the investigated operating situations.

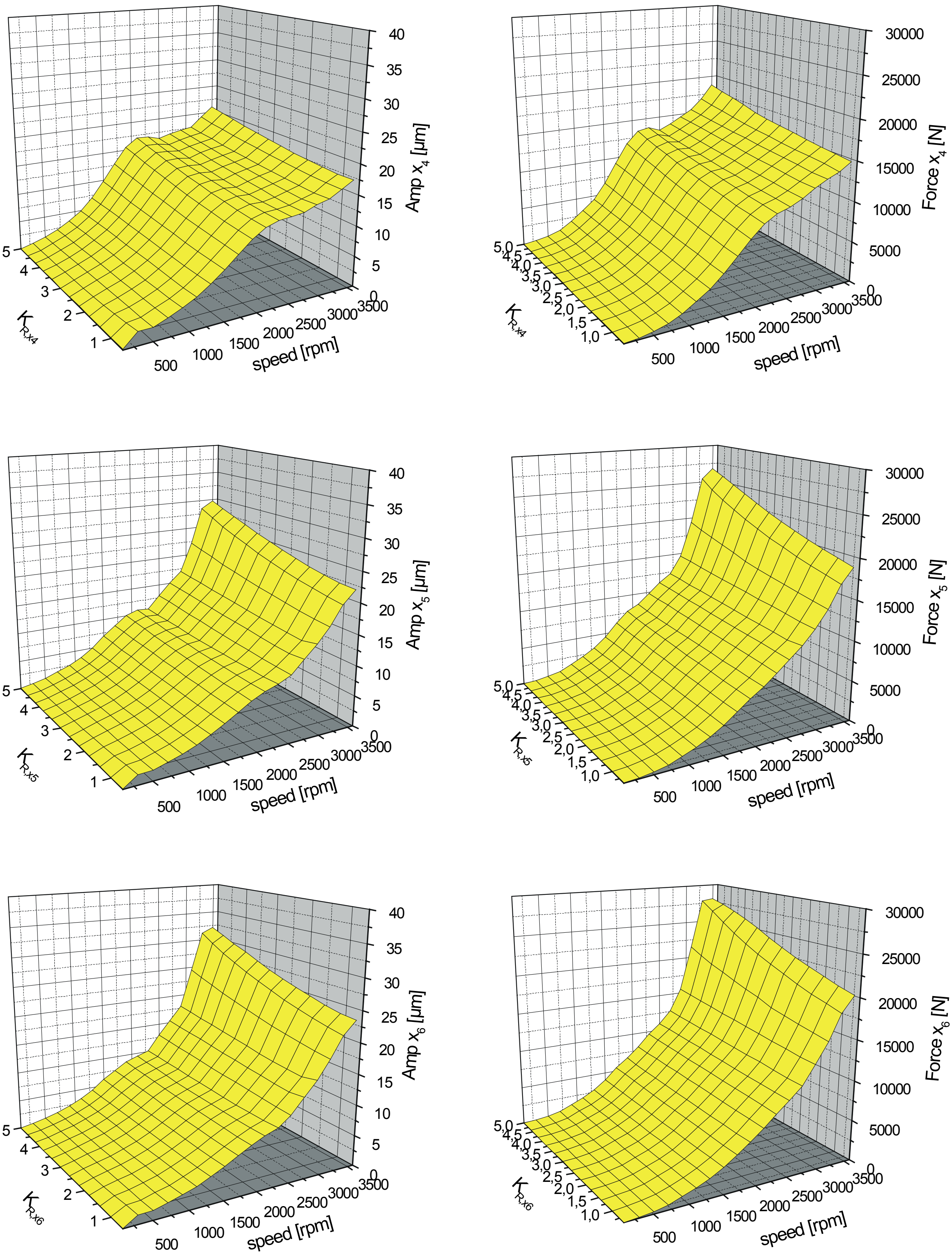
The axial AMB needs to be further optimized regarding the time constants of the coils (inductivity) to meet the criteria of the DDS method. A way to decrease the time constants can be the development and usage of more powerful power amplifiers (higher closed loop voltage).

Furthermore it is necessary to design the axial AMB with upper and lower coils since the aerodynamical forces are acting in both directions depending on the speed of the turbomachine shaft.

Annex 1 – Speed levels and unbalances

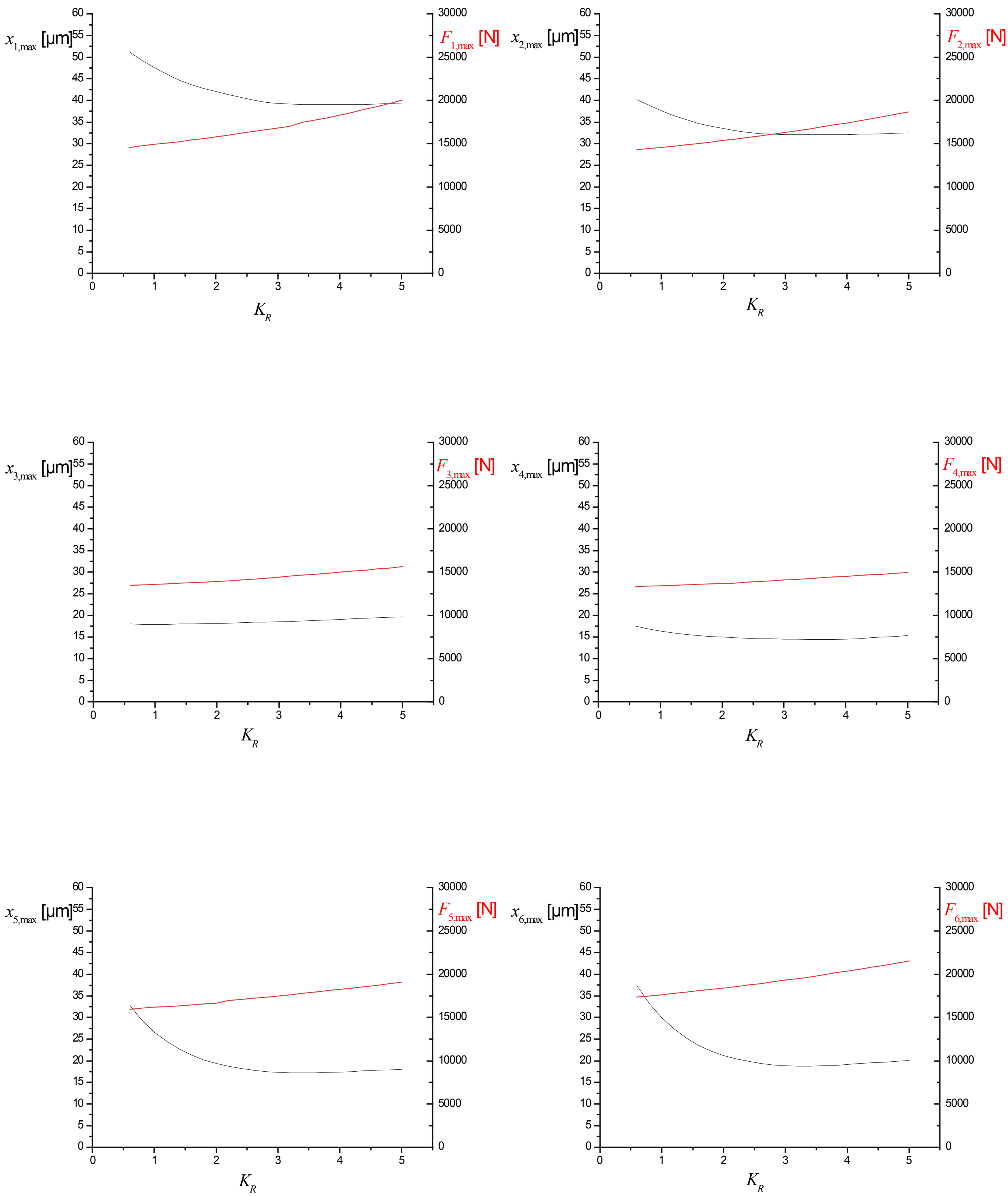


Amplitude of rotor position and bearing force vs. speed and controller gain for radial AMB 1...3



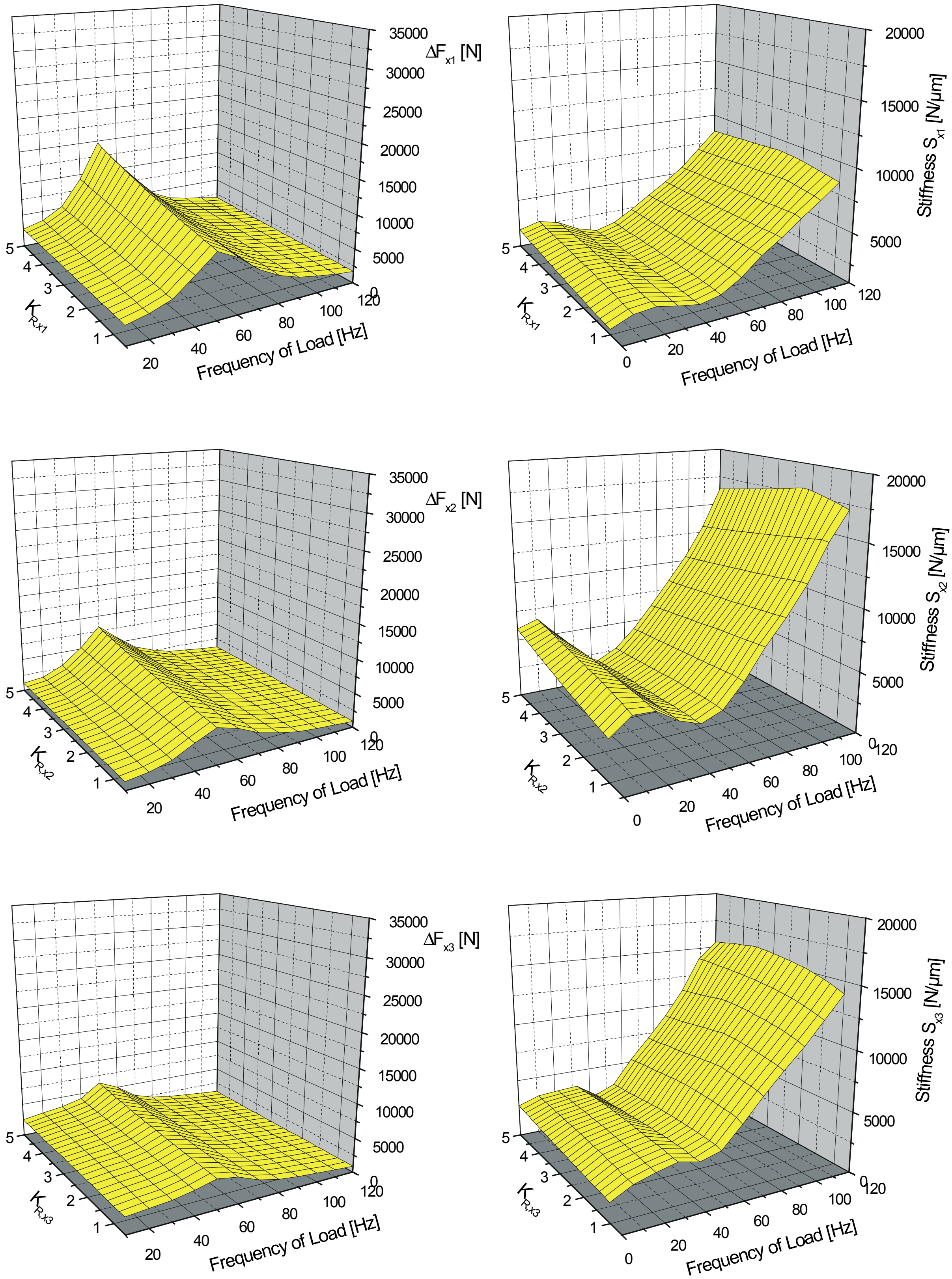
Amplitude of rotor position and bearing force vs. speed and controller gain for radial AMB 4...6

Annex 2 - Static loads at radial AMB's

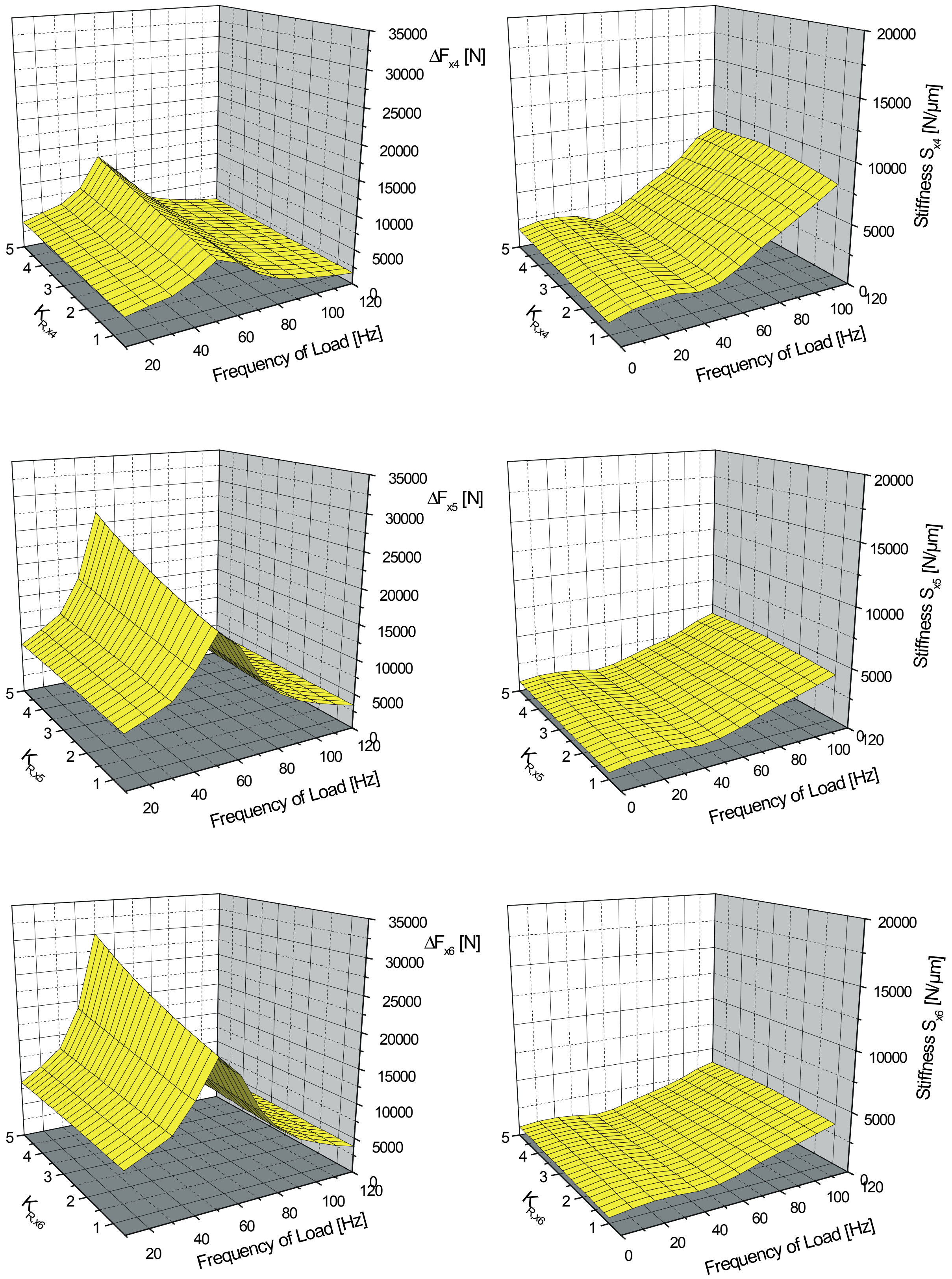


Amplitude of Rotor Position and Bearing Force vs. Controller Gain for radial AMB 1...6

Annex 3 – Dynamic loads at radial AMB's

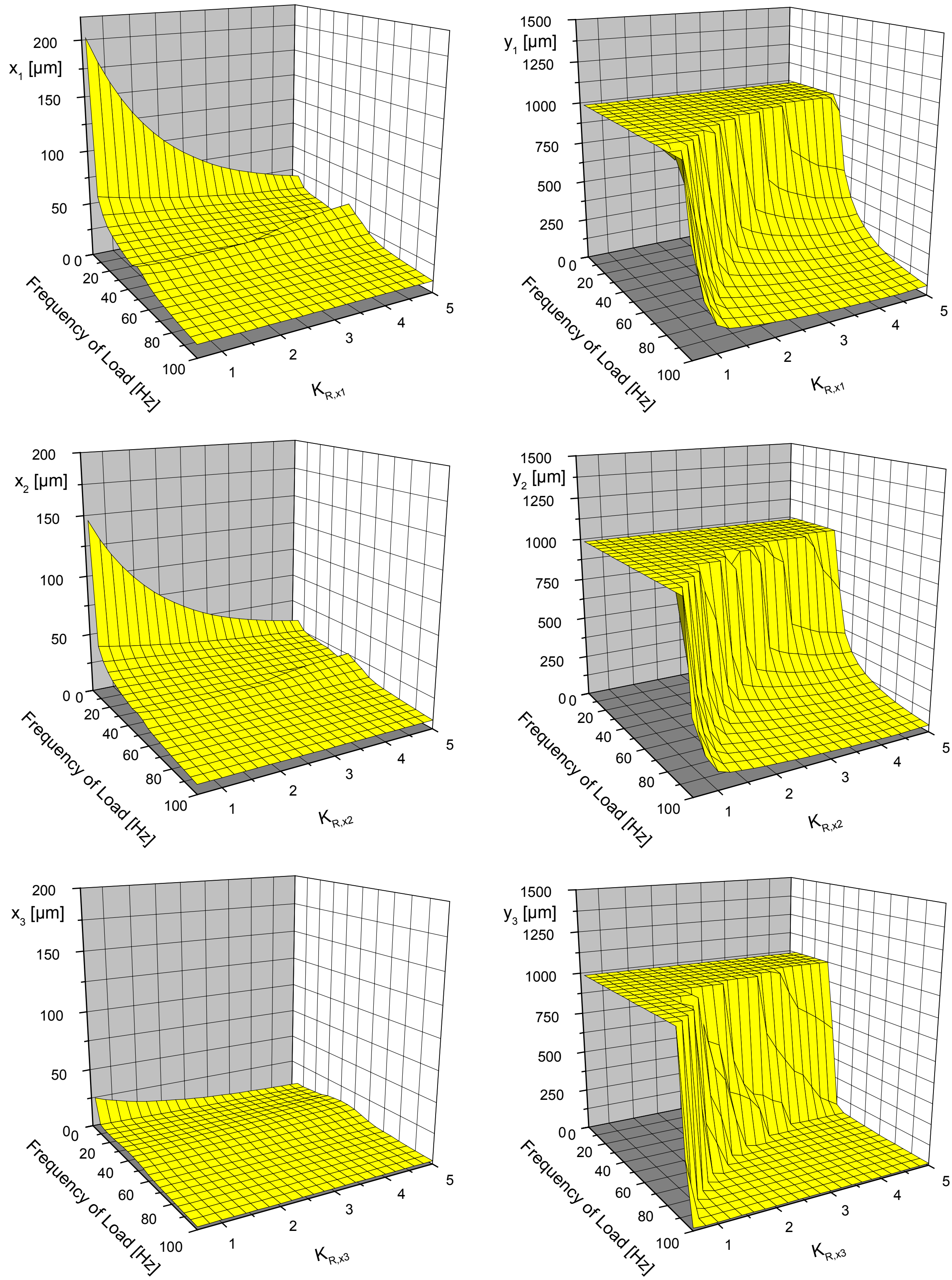


Bearing Force and Dynamic Stiffness vs. Freqency of Load and Controller Gain for radial AMB 1...3

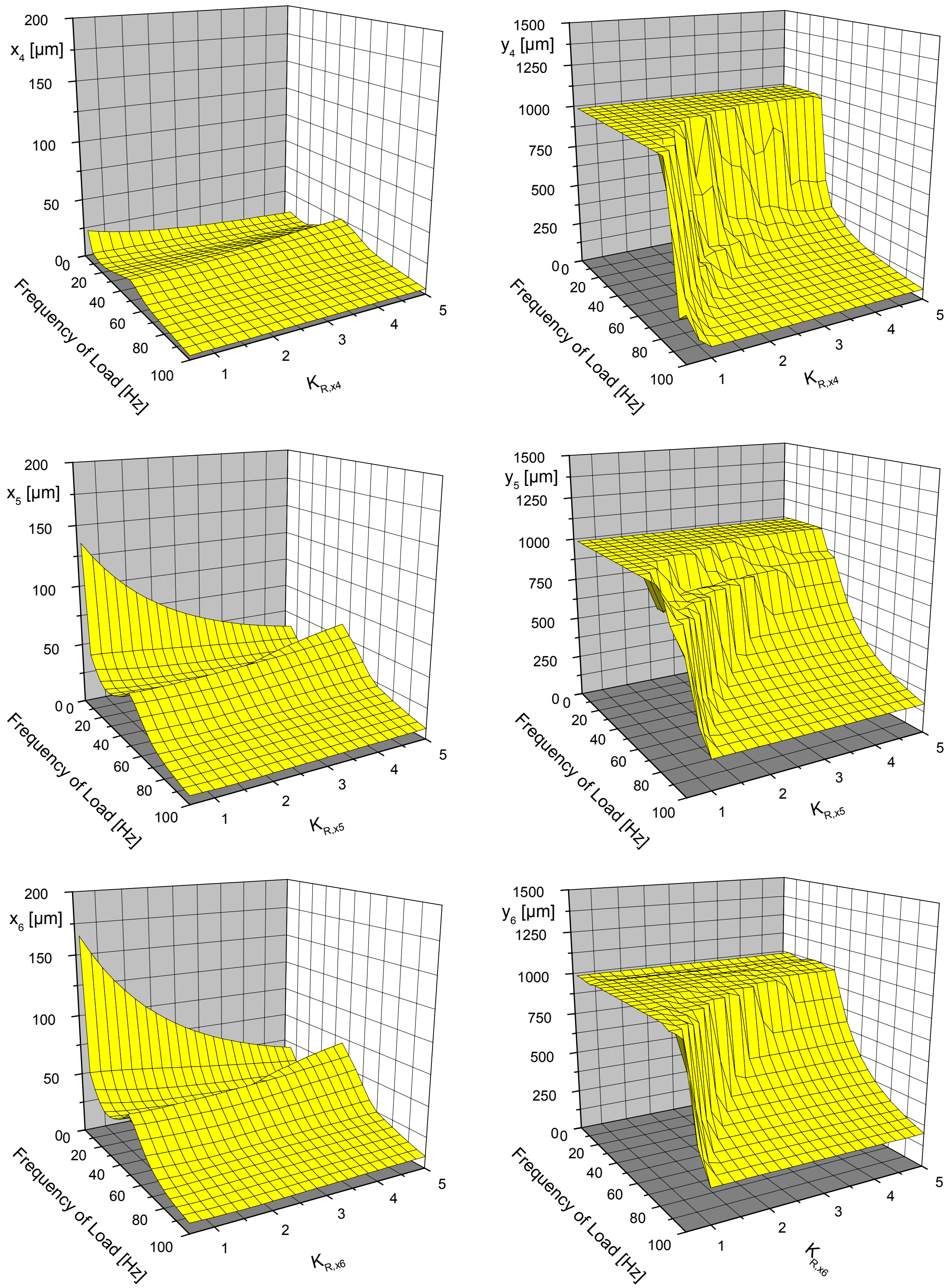


Bearing Force and Dynamic Stiffness vs. Freqency of Load and Controller Gain for radial AMB 4...6

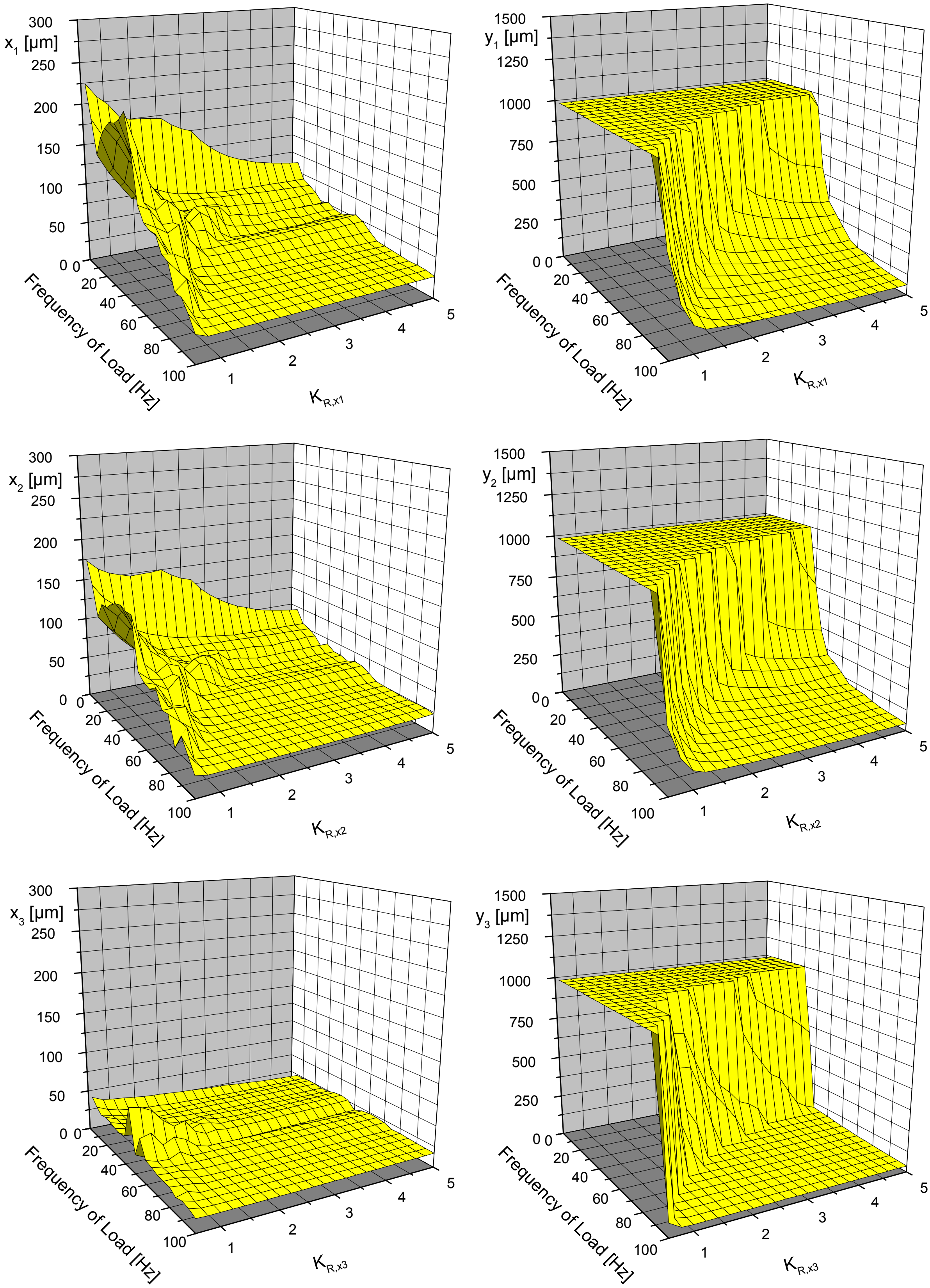
Annex 4 – Seismic loads at radial AMB's



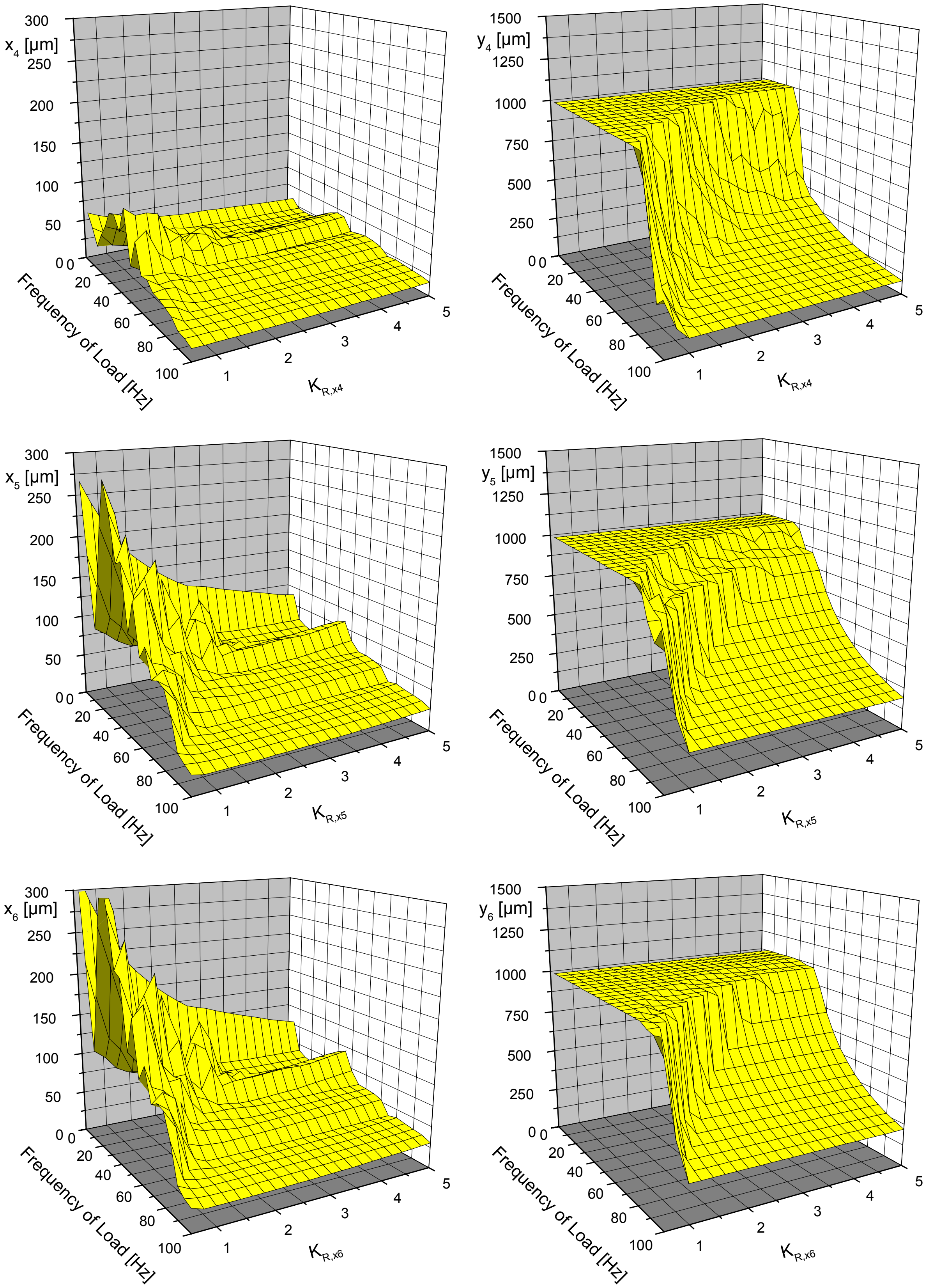
Amplitude of Rotor Position in x- and y-Direction vs. Freqency of Load and Controller Gain for radial AMB 1...3 at rotational speed 0 rpm



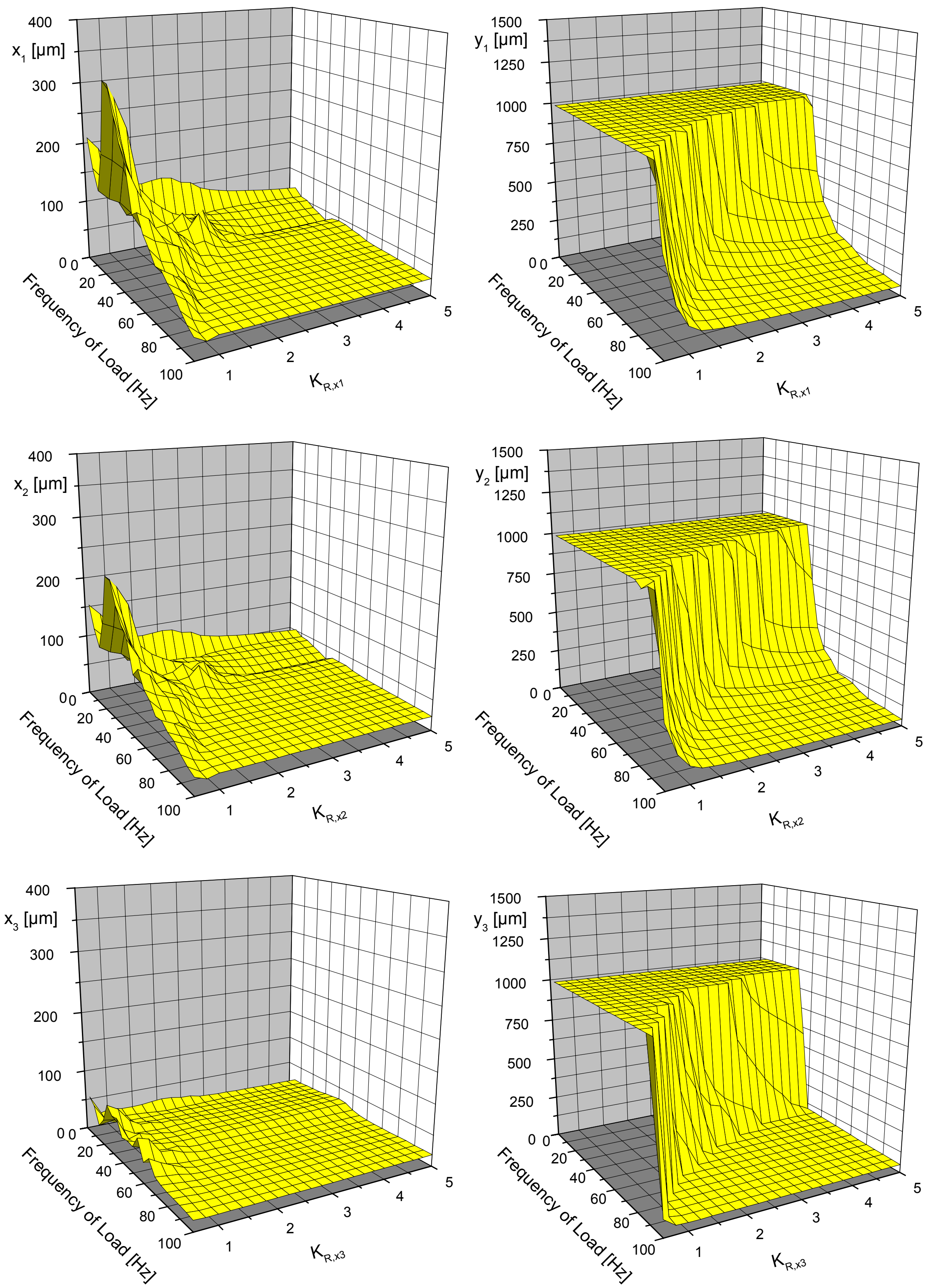
Amplitude of Rotor Position in x- and y-Direction vs. Freqency of Load and Controller Gain for radial AMB 4...6 at rotational speed 0 rpm



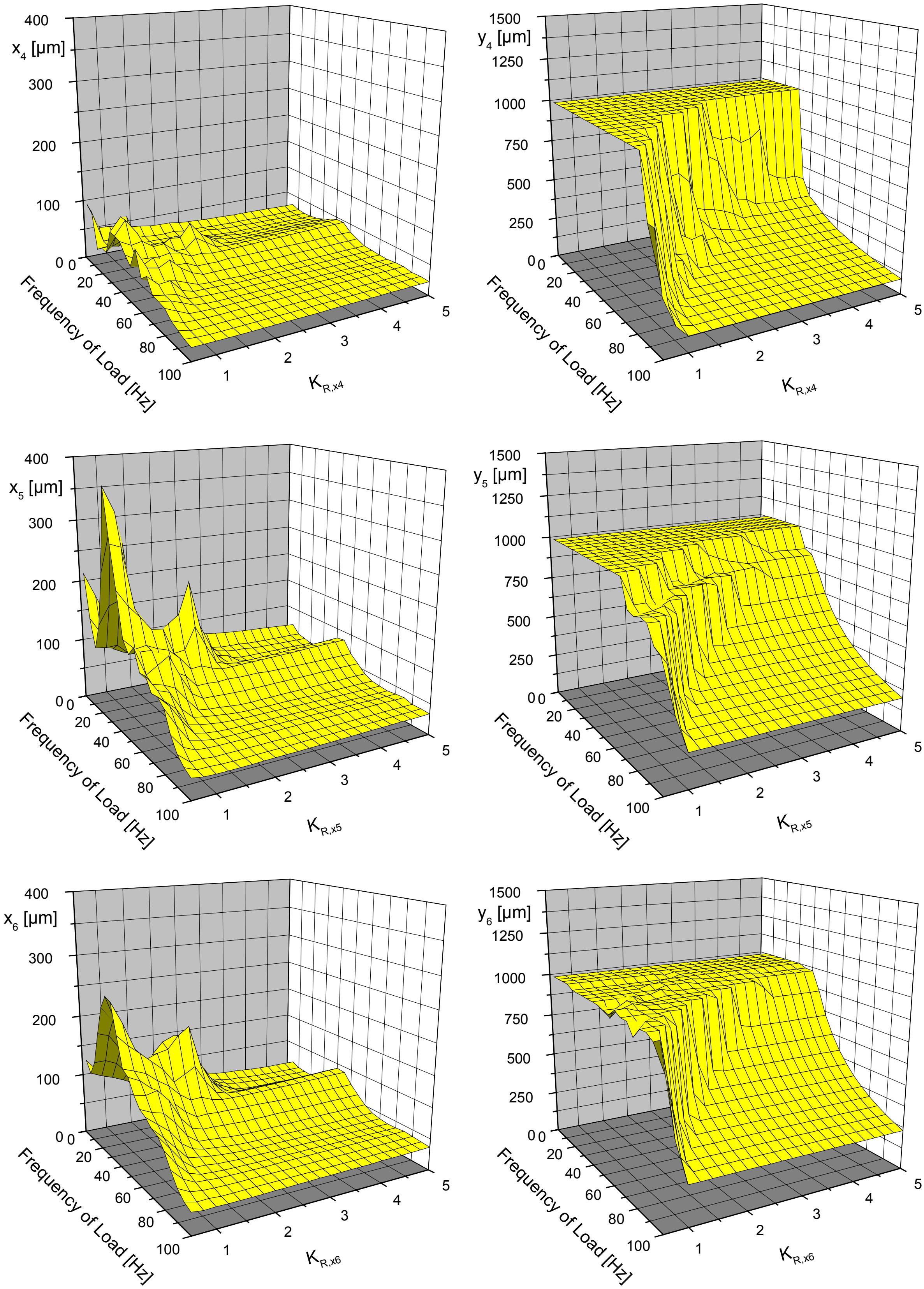
Amplitude of Rotor Position in x- and y-Direction vs. Freqency of Load and Controller Gain for radial AMB 1...3 at rotational speed 3000 rpm



Amplitude of Rotor Position in x- and y-Direction vs. Freqency of Load and Controller Gain for radial AMB 4...6 at rotational speed 3000 rpm



Amplitude of Rotor Position in x- and y-Direction vs. Freqency of Load and Controller Gain for radial AMB 1...3 at rotational speed 3600 rpm



Amplitude of Rotor Position in x- and y-Direction vs. Freqency of Load and Controller Gain for radial AMB 4...6 at rotational speed 3600 rpm